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U1S S1819 S1984

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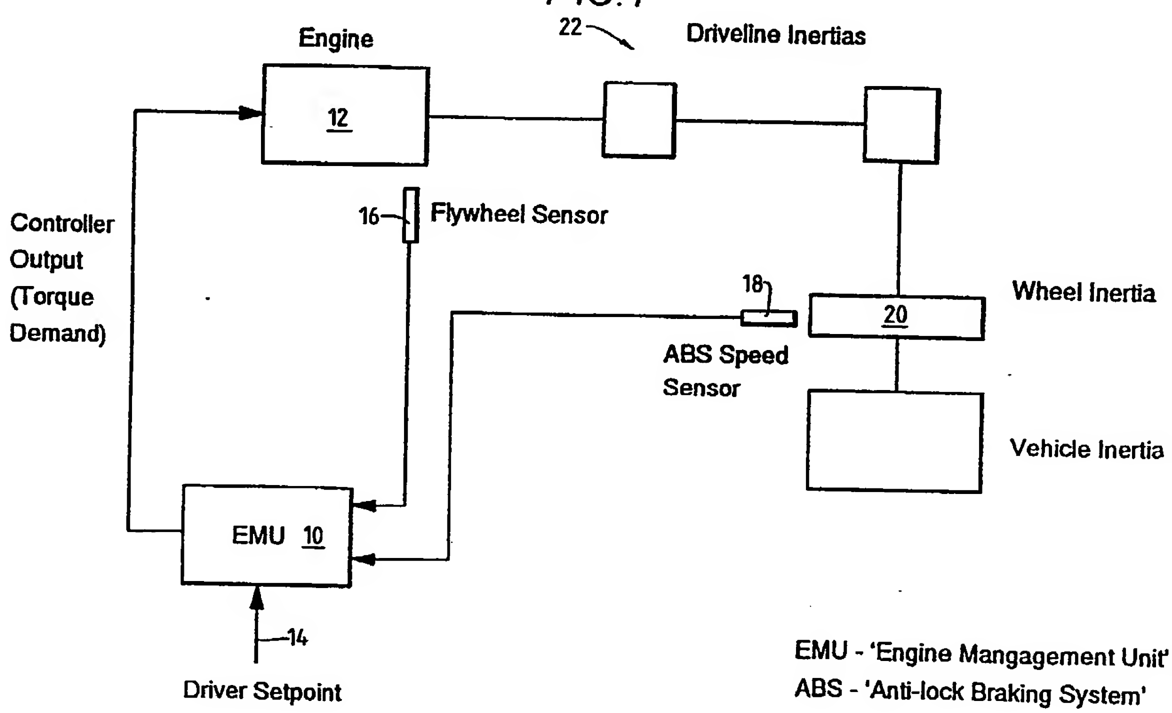
(54) Vibration reduced speed control

(57) An engine speed control system for reducing vibration comprises a control unit 10 programmed with a predictive vibration model of the drive system of the vehicle that outputs a torque control signal according to the speed or torque 14 requested by the driver.

Preferably vibration or velocity sensors eg ABS speed sensors 16 may feedback the actual response of the drive system to modify the predictive model.

Vibration is reduced without significantly the response time to demanded torque changes.

FIG. 1



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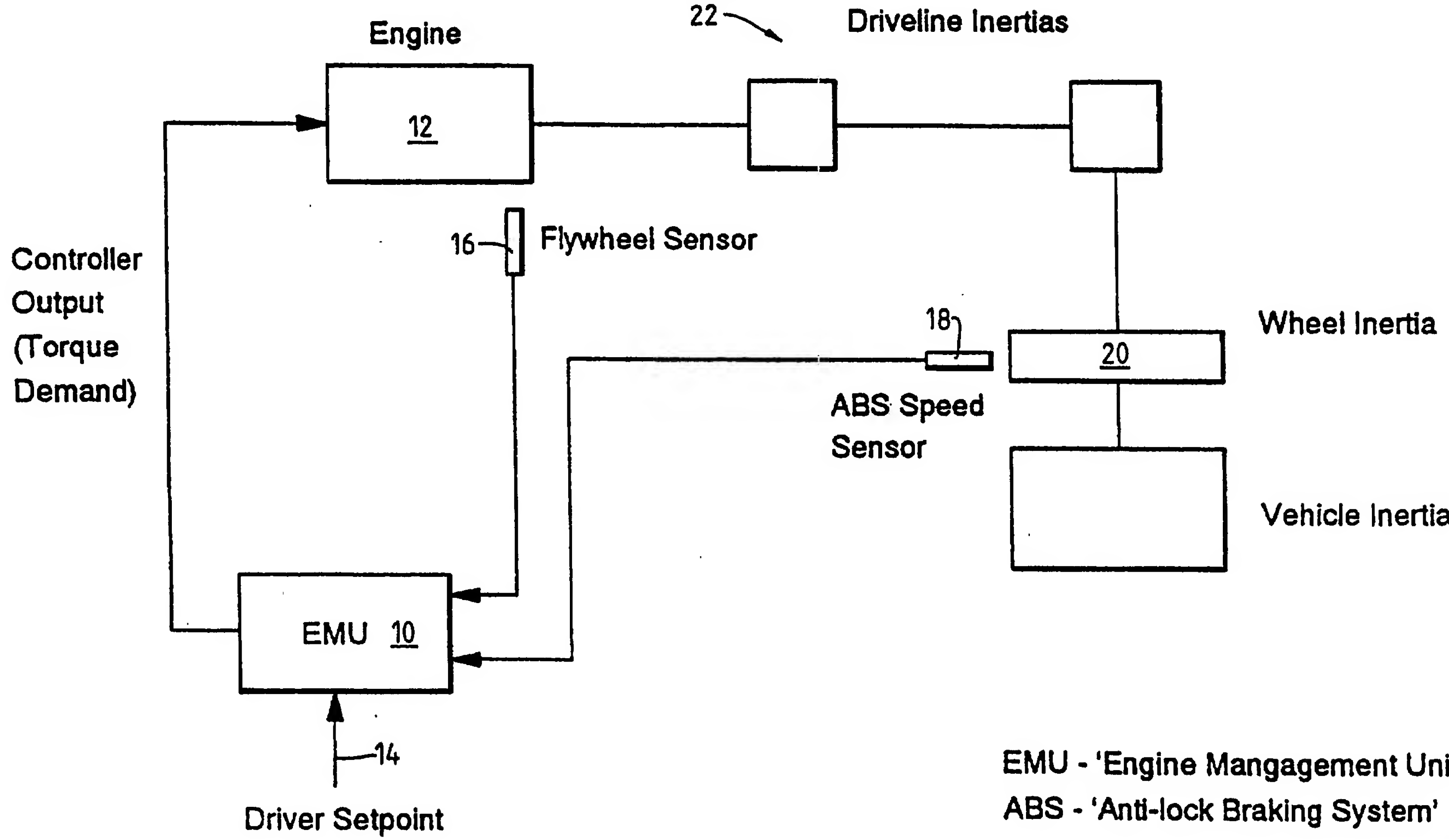
At least one drawing originally filed was informal and the print reproduced here is taken from a later filed formal copy.

FIG. 1



EMU - 'Engine Mangagement Unit'
ABS - 'Anti-lock Braking System'

FIG. 2a

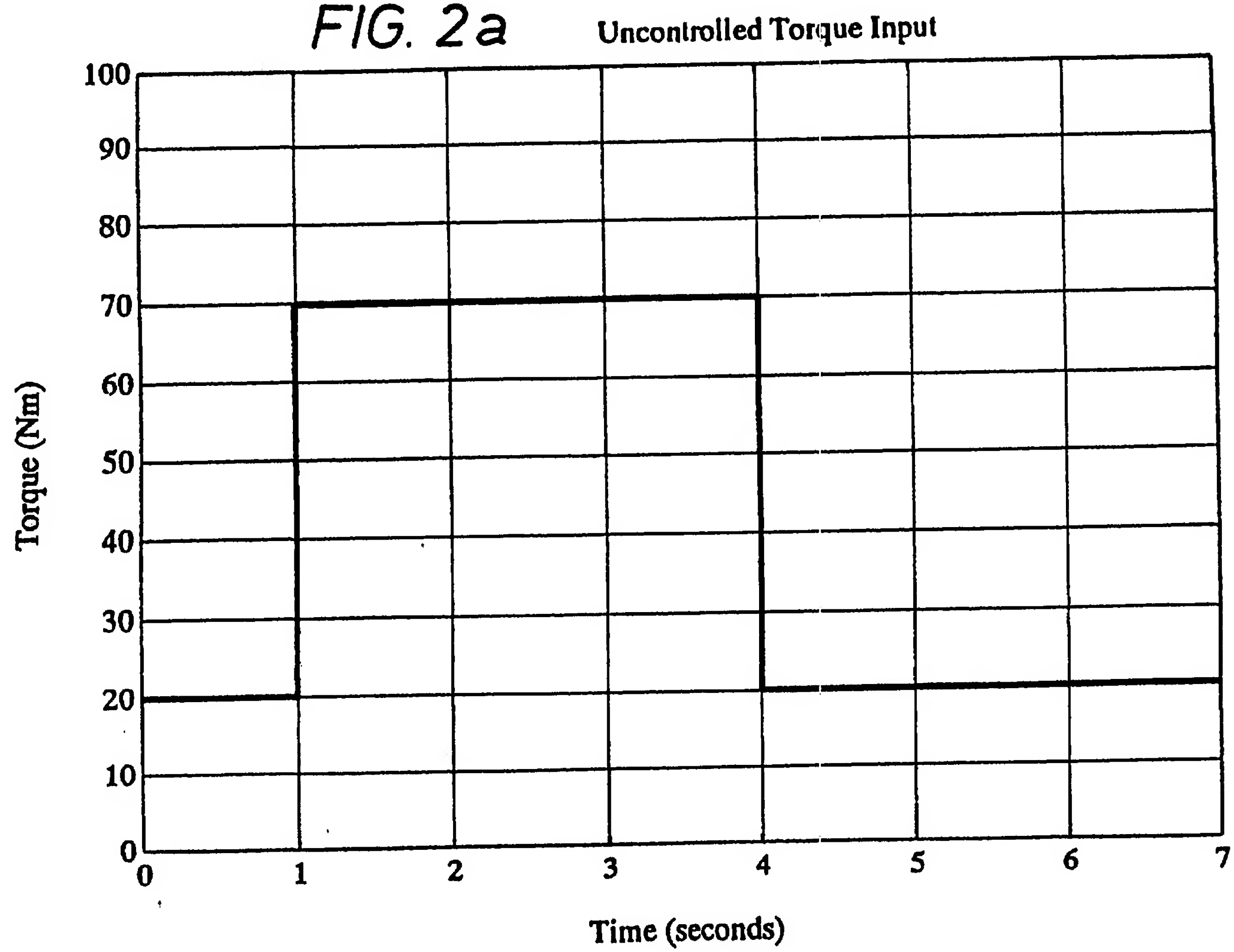

Uncontrolled Torque Input
Torque (Nm)
100
90
80
70
60
50
40
30
20
10
0
0
1
2
3
4
5
6
7
Time (seconds)

FIG. 2b

Open-Loop Response of Driveline Inertias

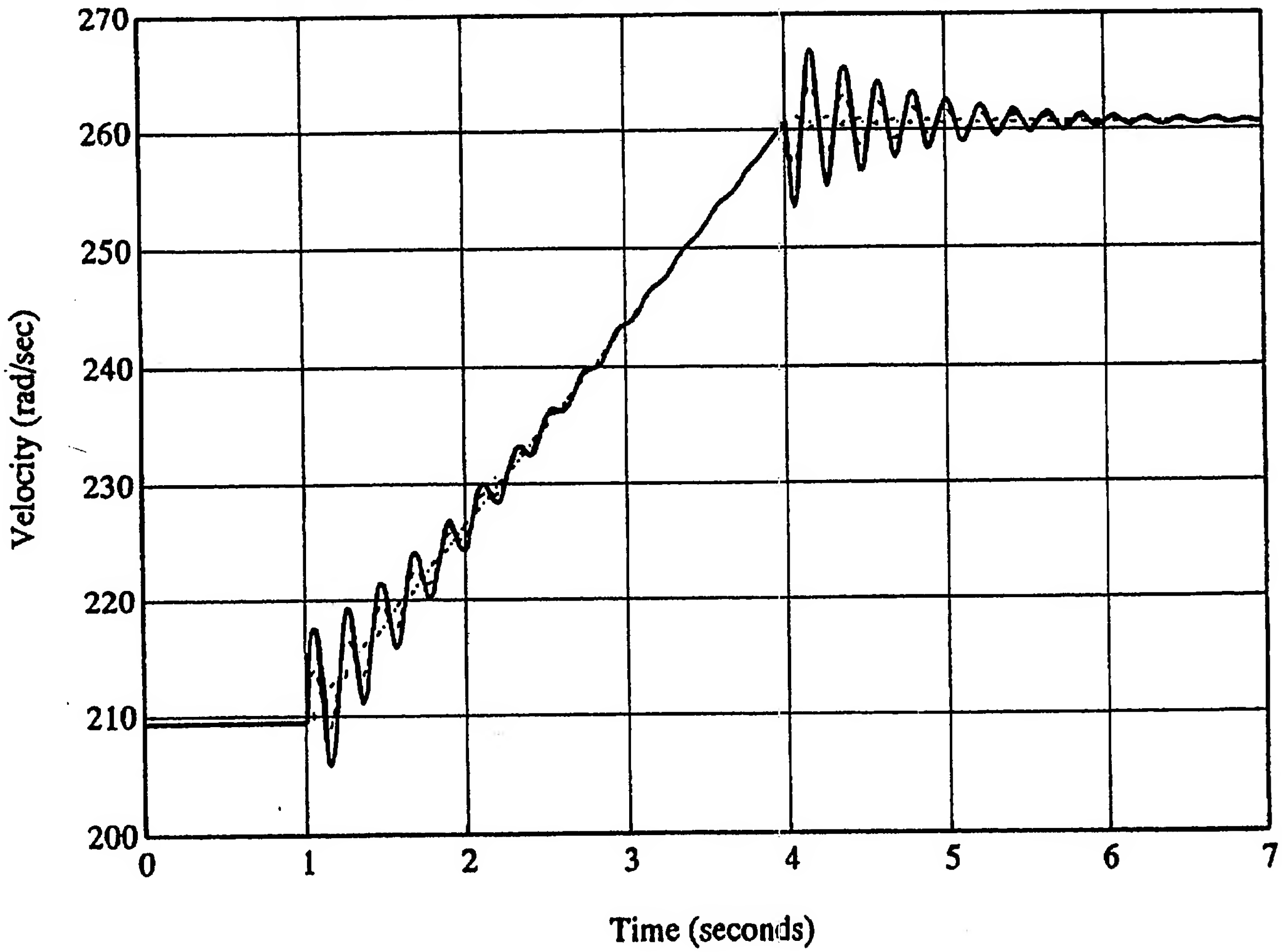


FIG. 3a

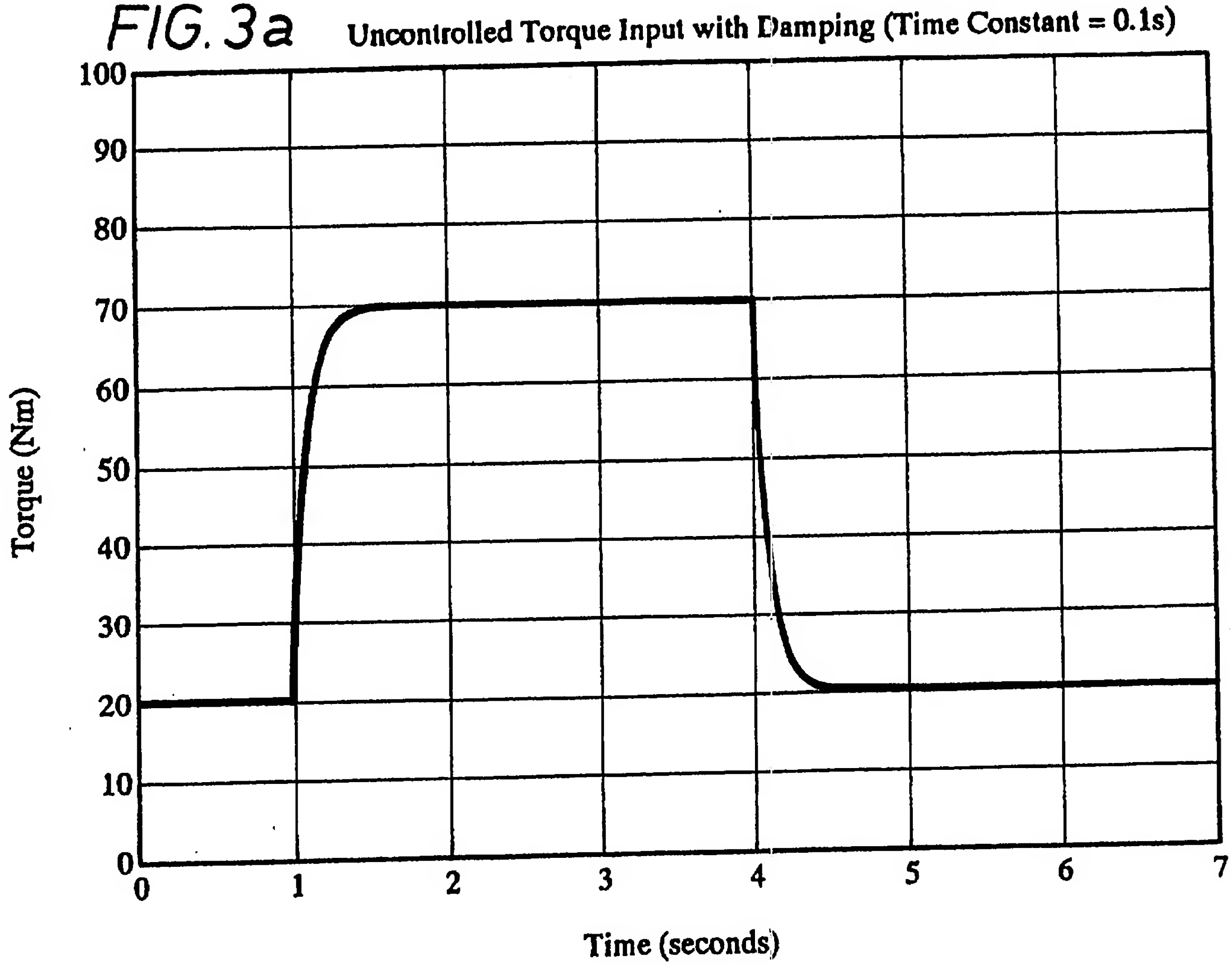

Uncontrolled Torque Input with Damping (Time Constant = 0.1s)
Torque (Nm)
100
90
80
70
60
50
40
30
20
10
0
0
1
2
3
4
5
6
7
Time (seconds)

FIG. 3b

Open-Loop Response of Driveline Inertias

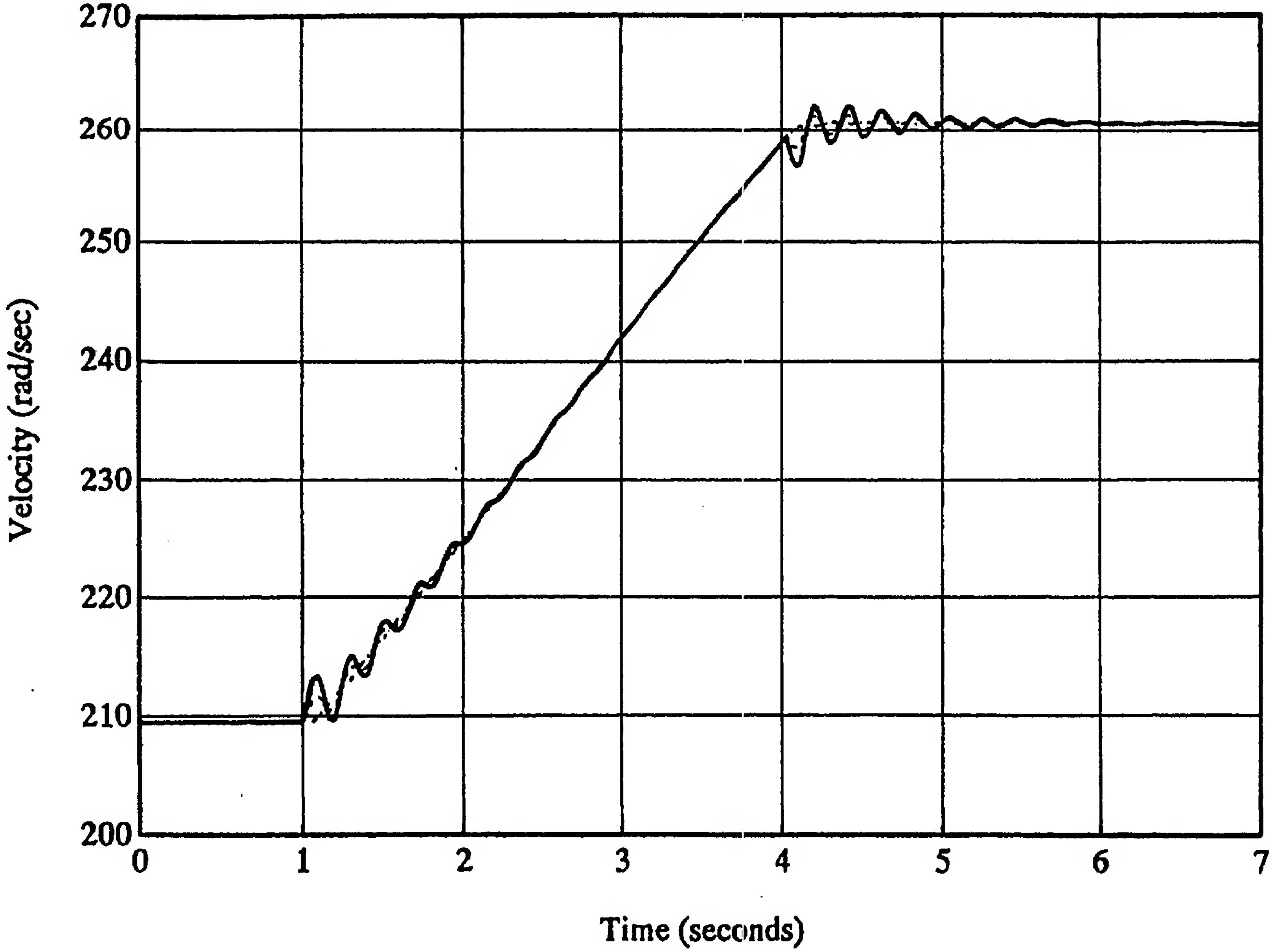


FIG. 4a Uncontrolled Torque Input with Damping (Time Constant = 0.6s)

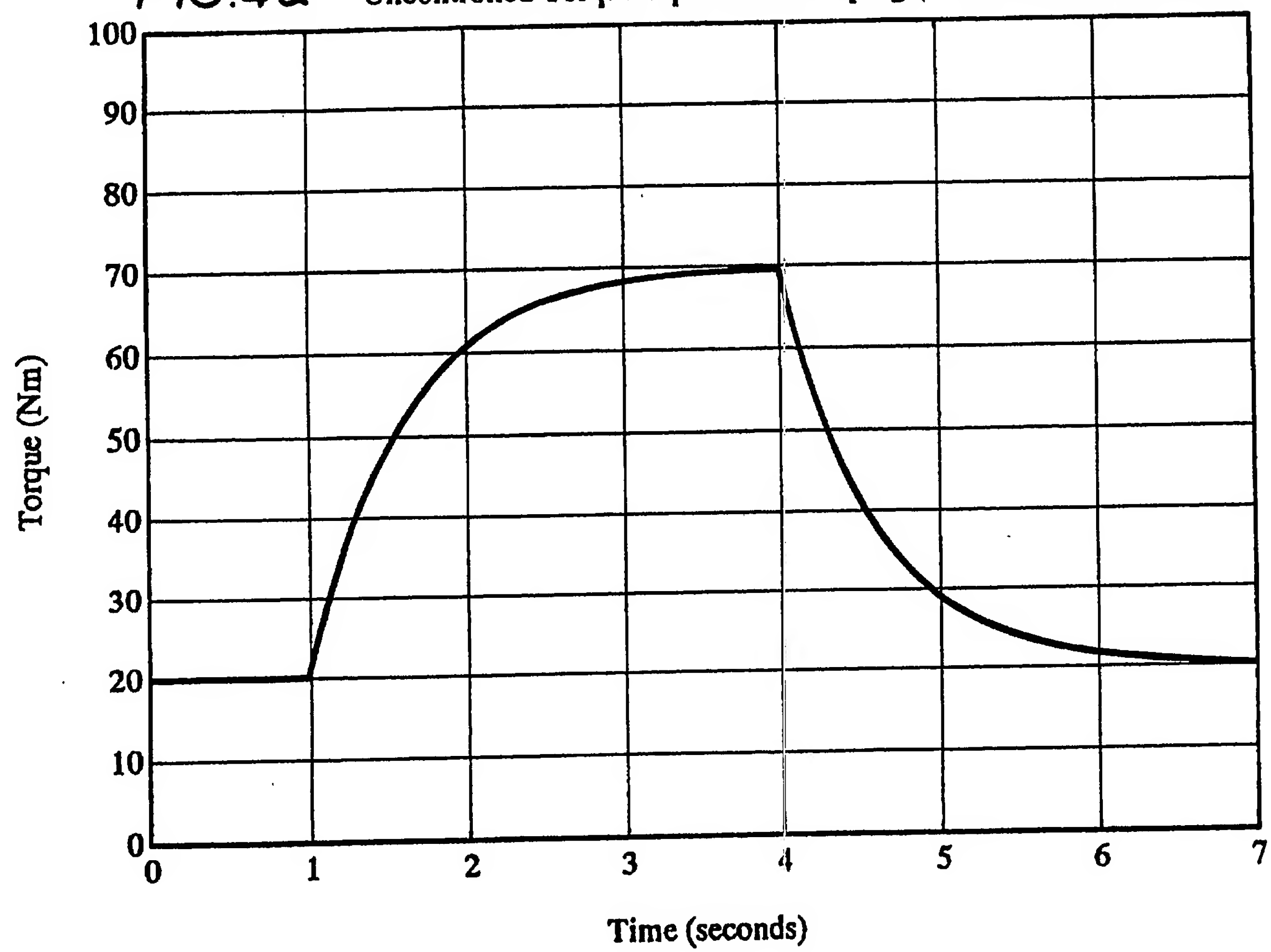


FIG. 4b

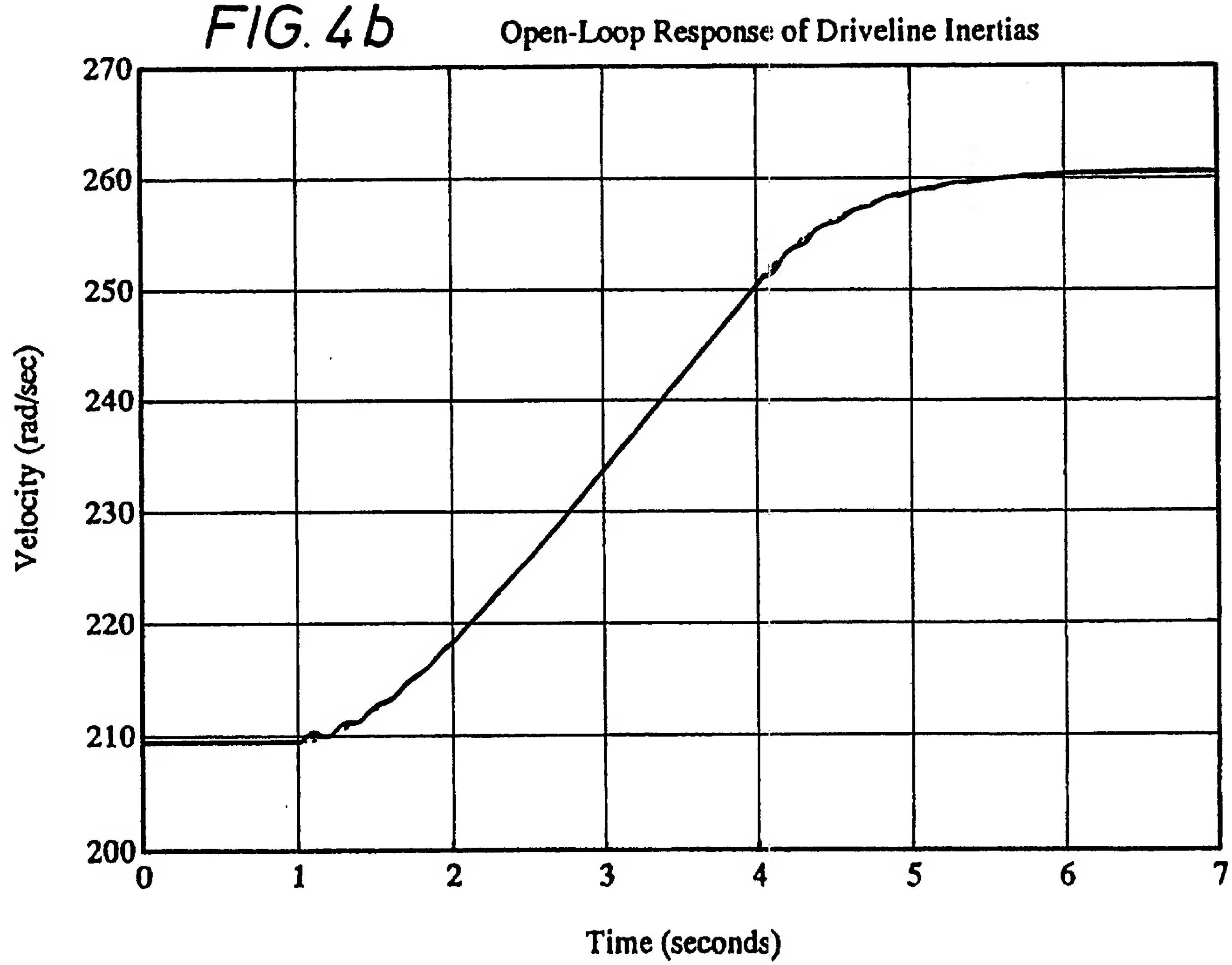

Open-Loop Response of Driveline Inertias
Velocity (rad/sec)
270
260
250
240
230
220
210
200
0
1
2
3
4
5
6
7
Time (seconds)

FIG. 5a Uncontrolled Torque Input following Sudden Clutch Engagement

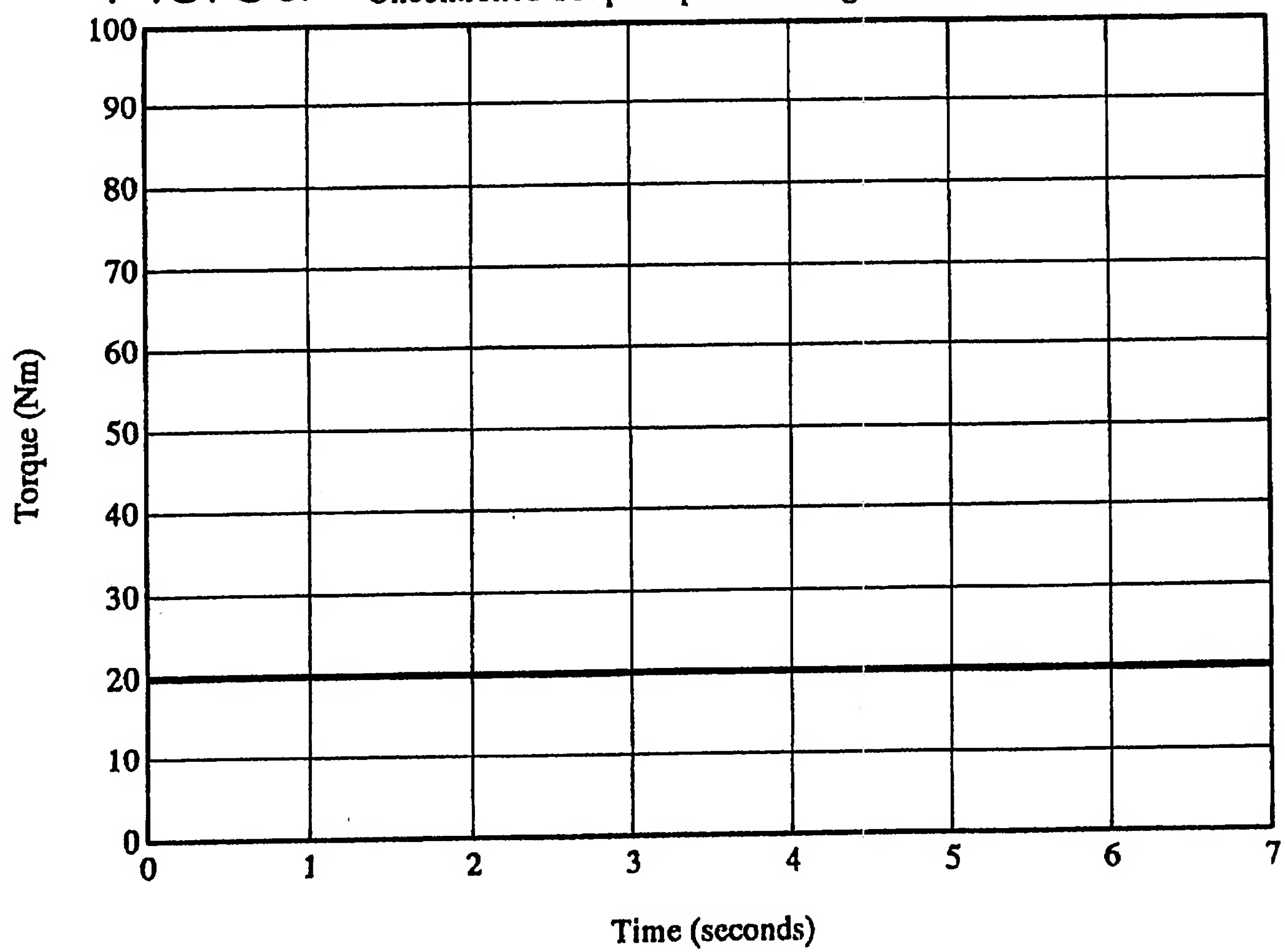
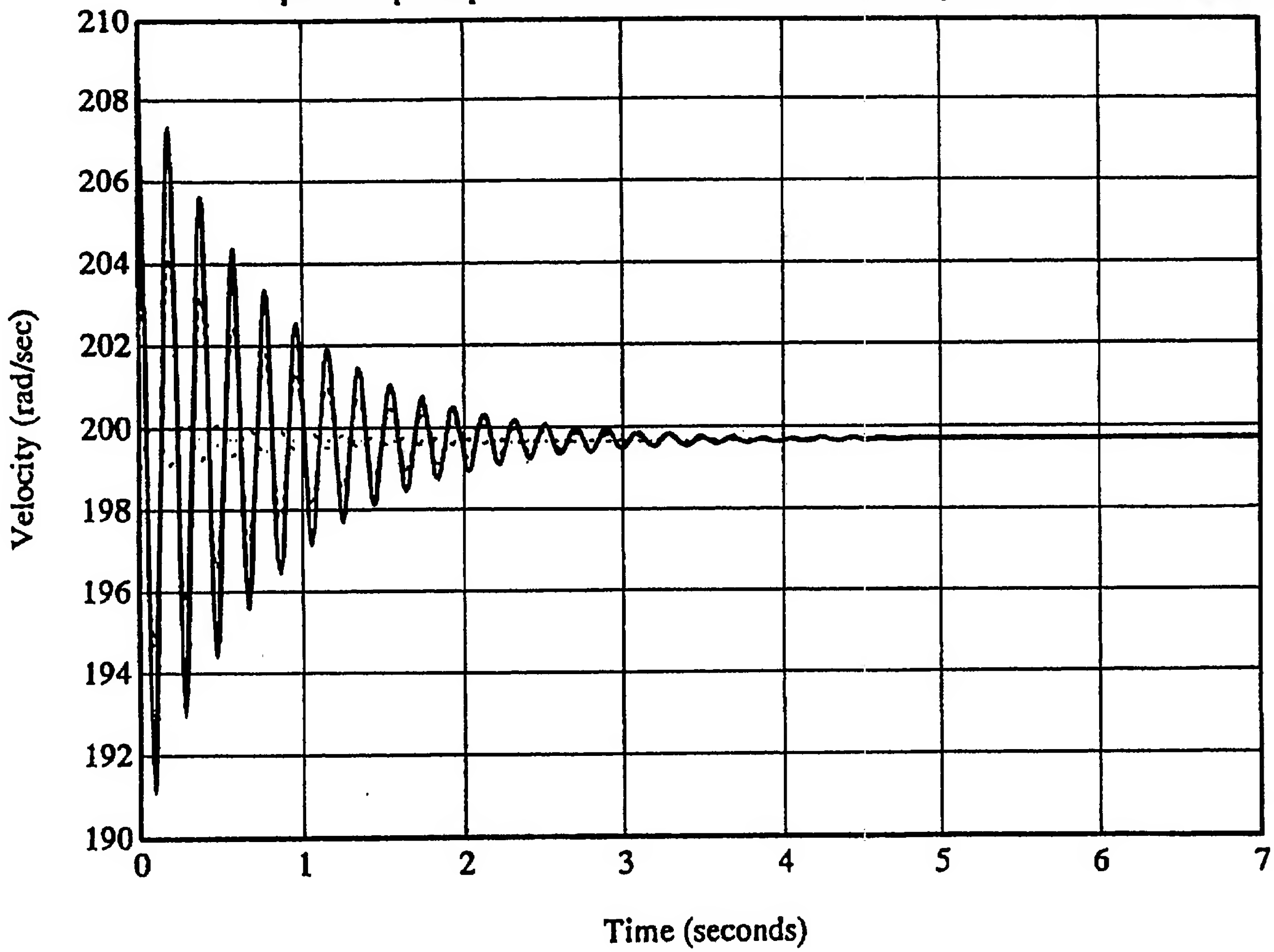


FIG. 5b

Open-Loop Response of Driveline Inertias following Sudden Clutch Engagement



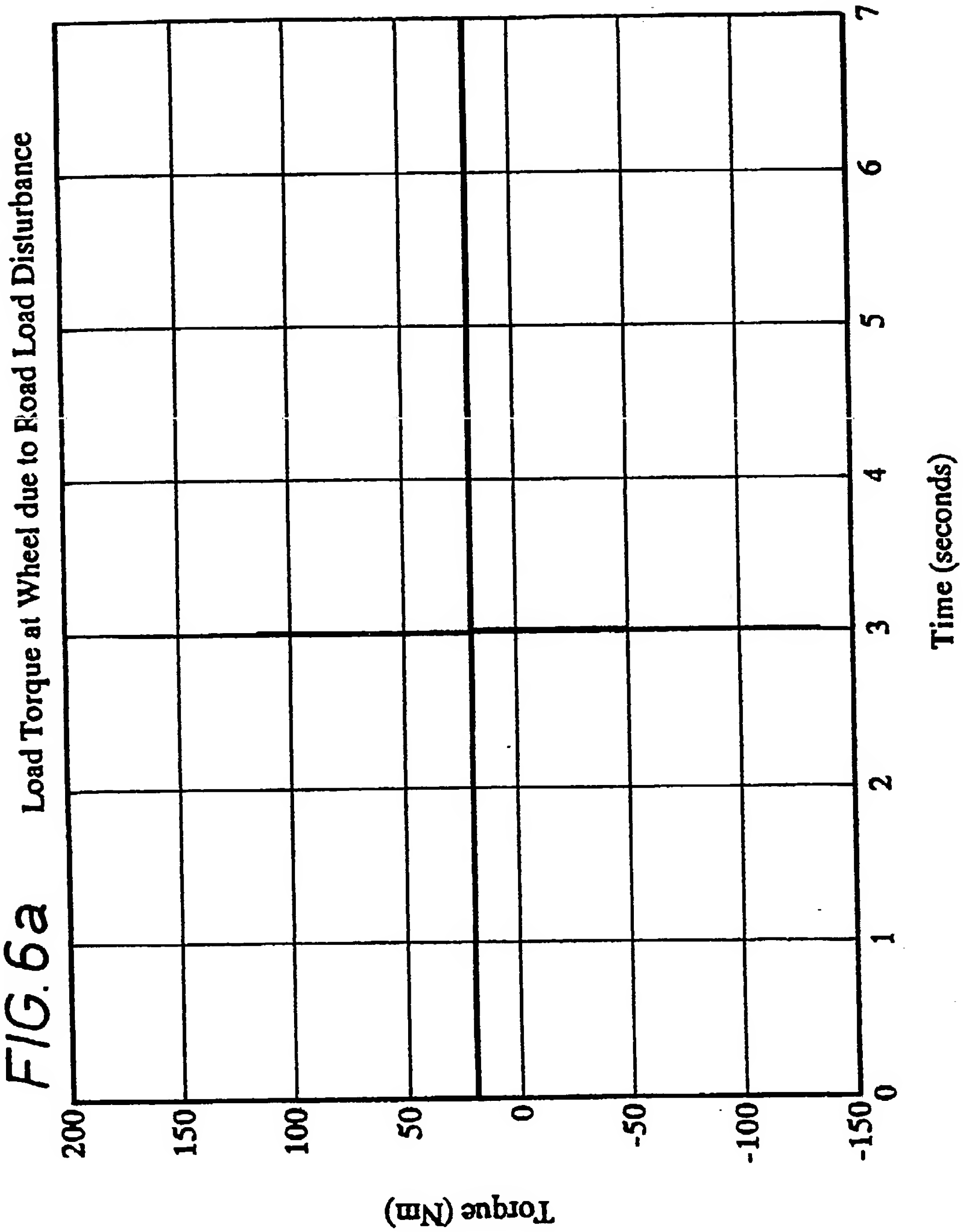

Load Torque at Wheel due to Road Load Disturbance
Torque (Nm)
200
150
100
50
0
-50
-100
-150
Time (seconds)
0
1
2
3
4
5
6
7

FIG. 6a

FIG.6b Open-Loop Response of Driveline Inertias to Road Load Disturbance

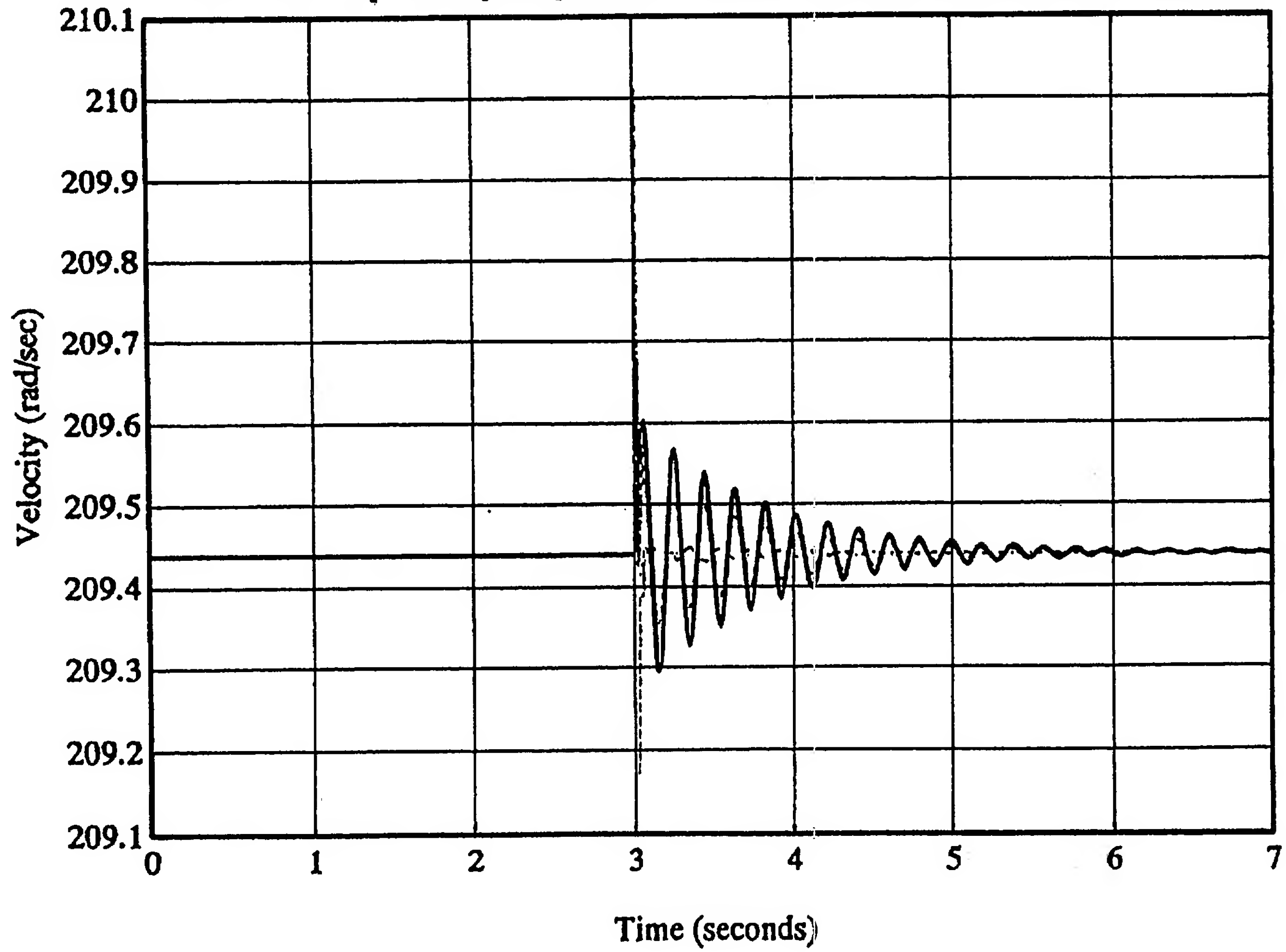


FIG. 7a

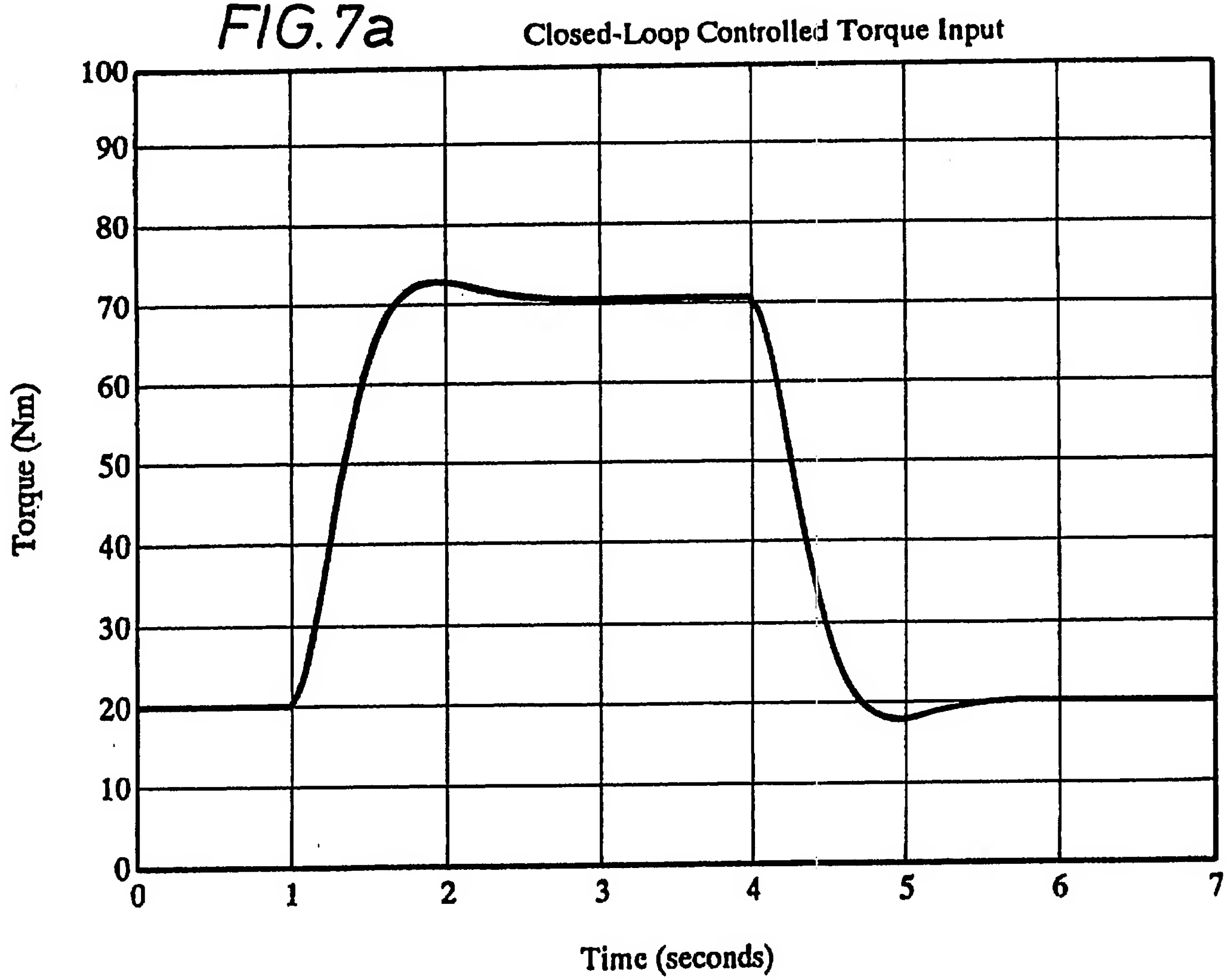

Closed-Loop Controlled Torque Input
Torque (Nm)
100
90
80
70
60
50
40
30
20
10
0
0
1
2
3
4
5
6
7
Time (seconds)

FIG. 7b

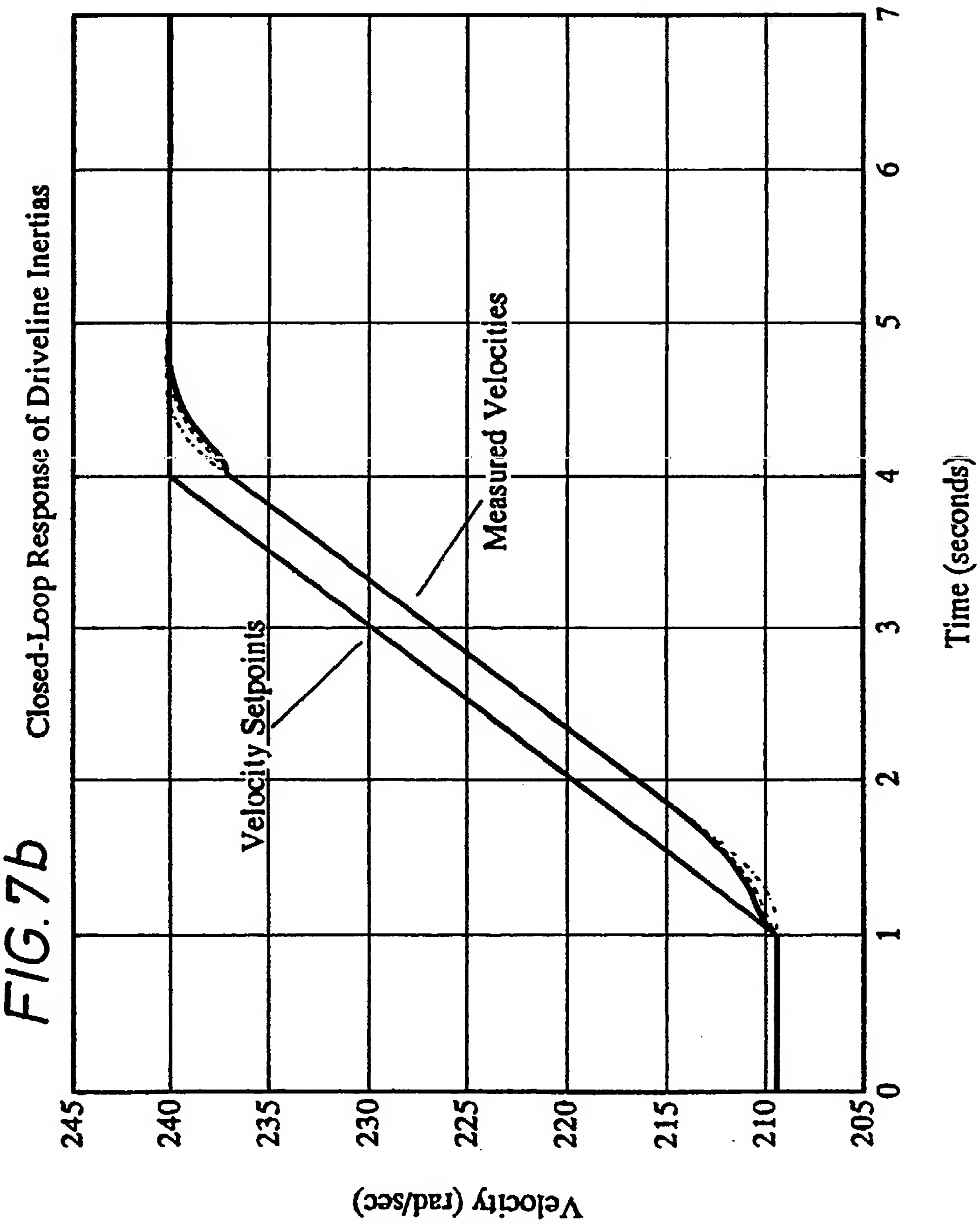

Closed-Loop Response of Driveline Inertias
Velocity Setpoints
Measured Velocities
Velocity (rad/sec)
245
240
235
230
225
220
215
210
205
Time (seconds)
0
1
2
3
4
5
6
7

FIG.8a Closed-Loop Controlled Torque Input

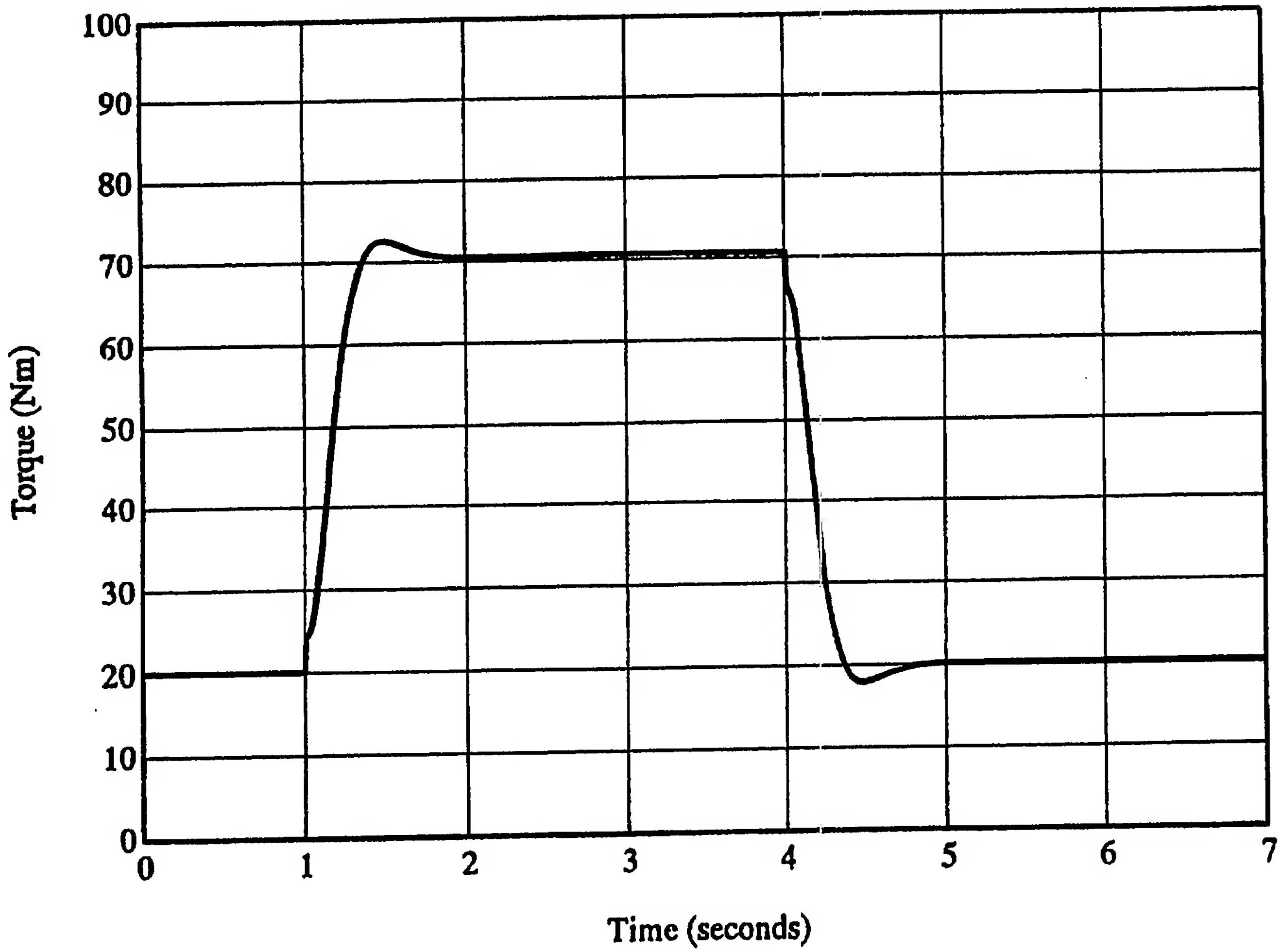


FIG. 8b

Closed-Loop Response of Driveline Inertias

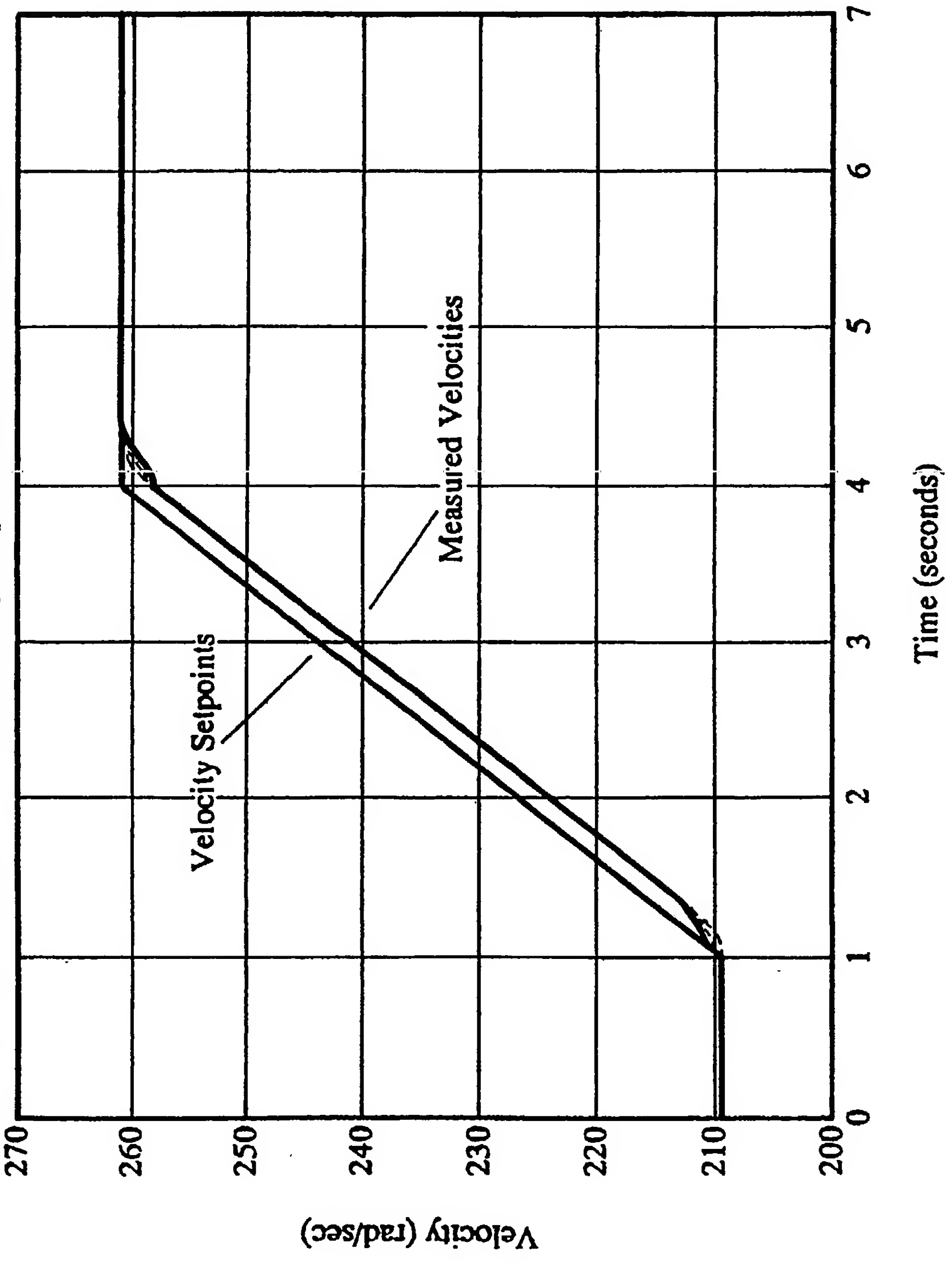


FIG. 9a

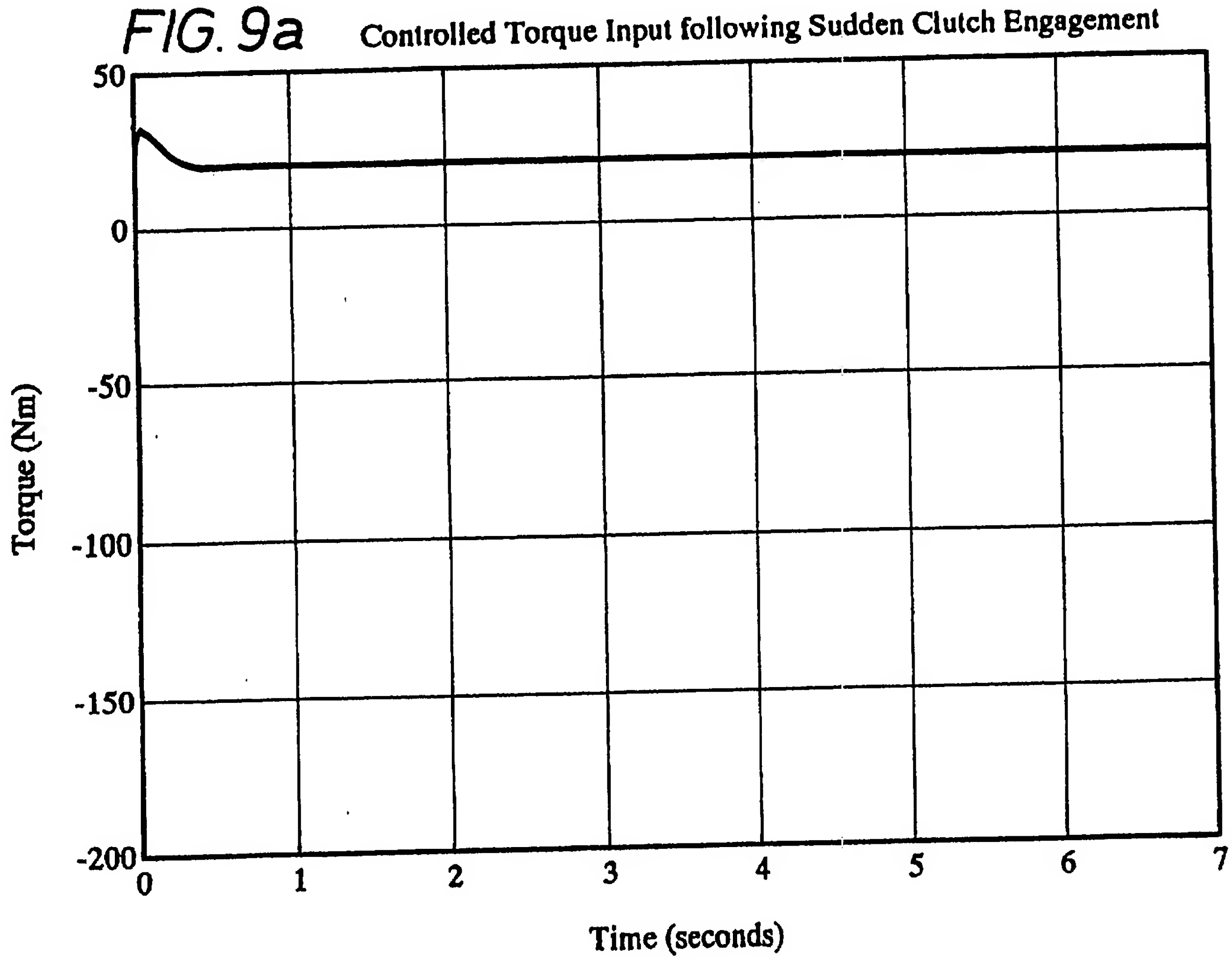

Controlled Torque Input following Sudden Clutch Engagement
50
0
-50
-100
-150
-200
Torque (Nm)
0
1
2
3
4
5
6
7
Time (seconds)

FIG. 9b Closed-Loop Response of Driveline Inertias following Sudden Clutch Engagement

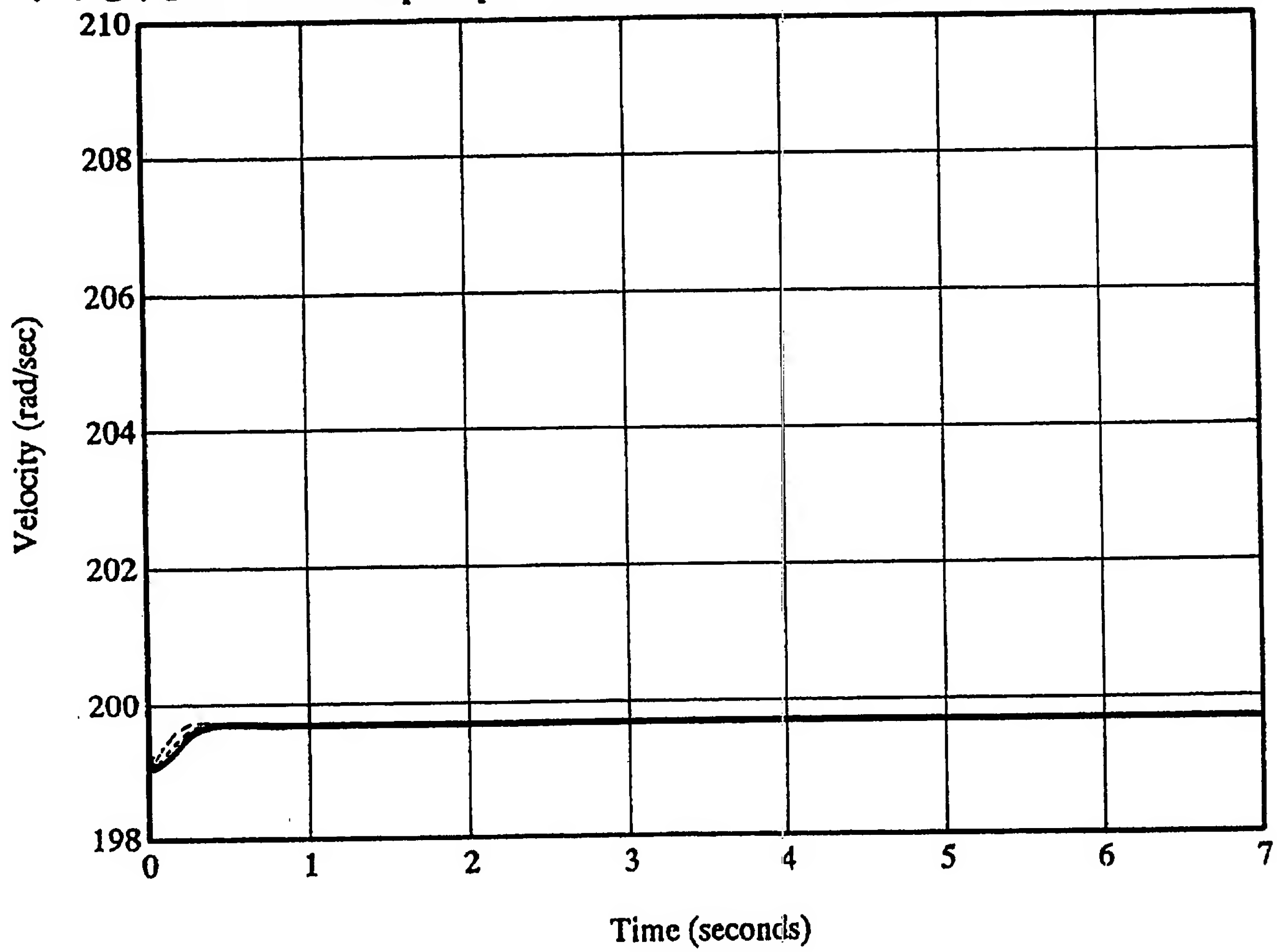


FIG.10a

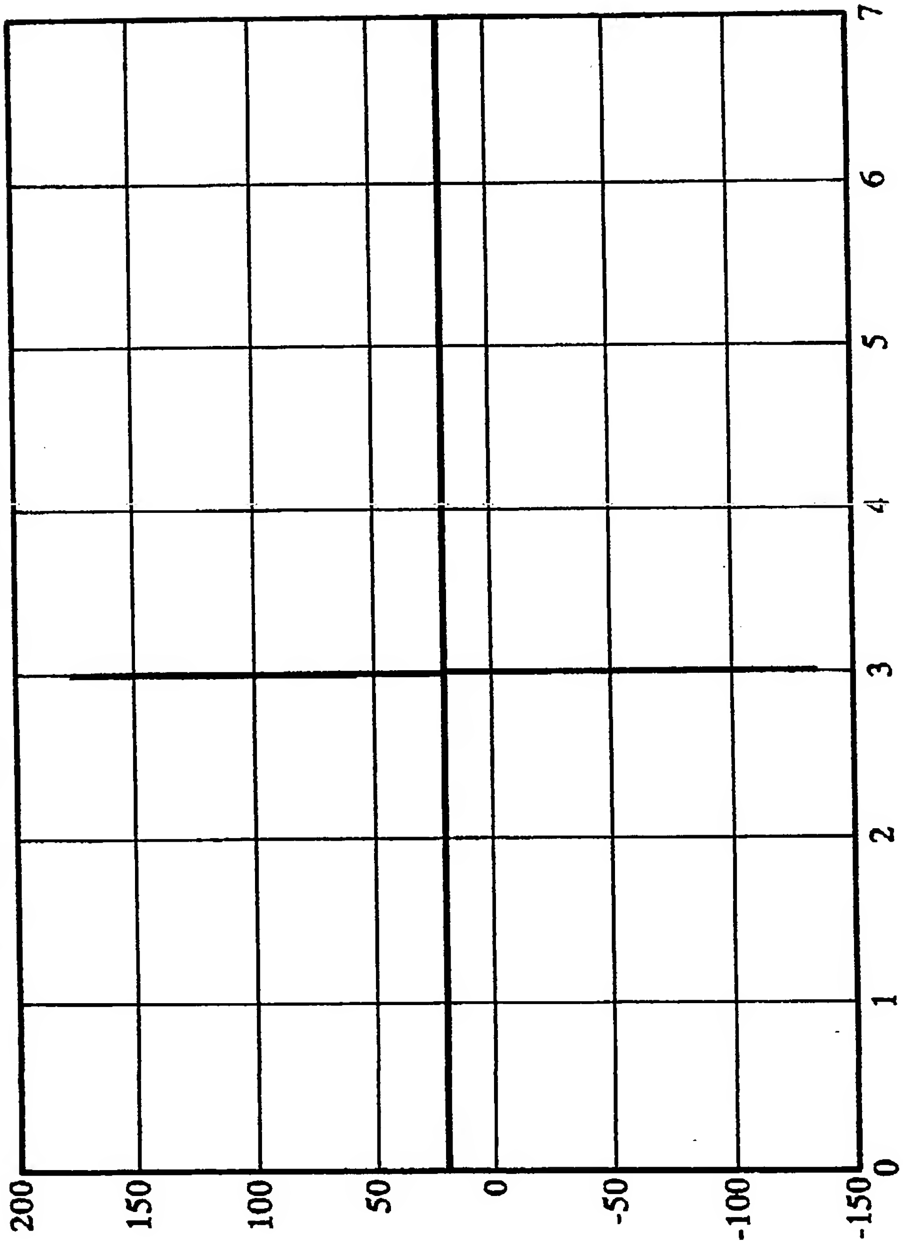
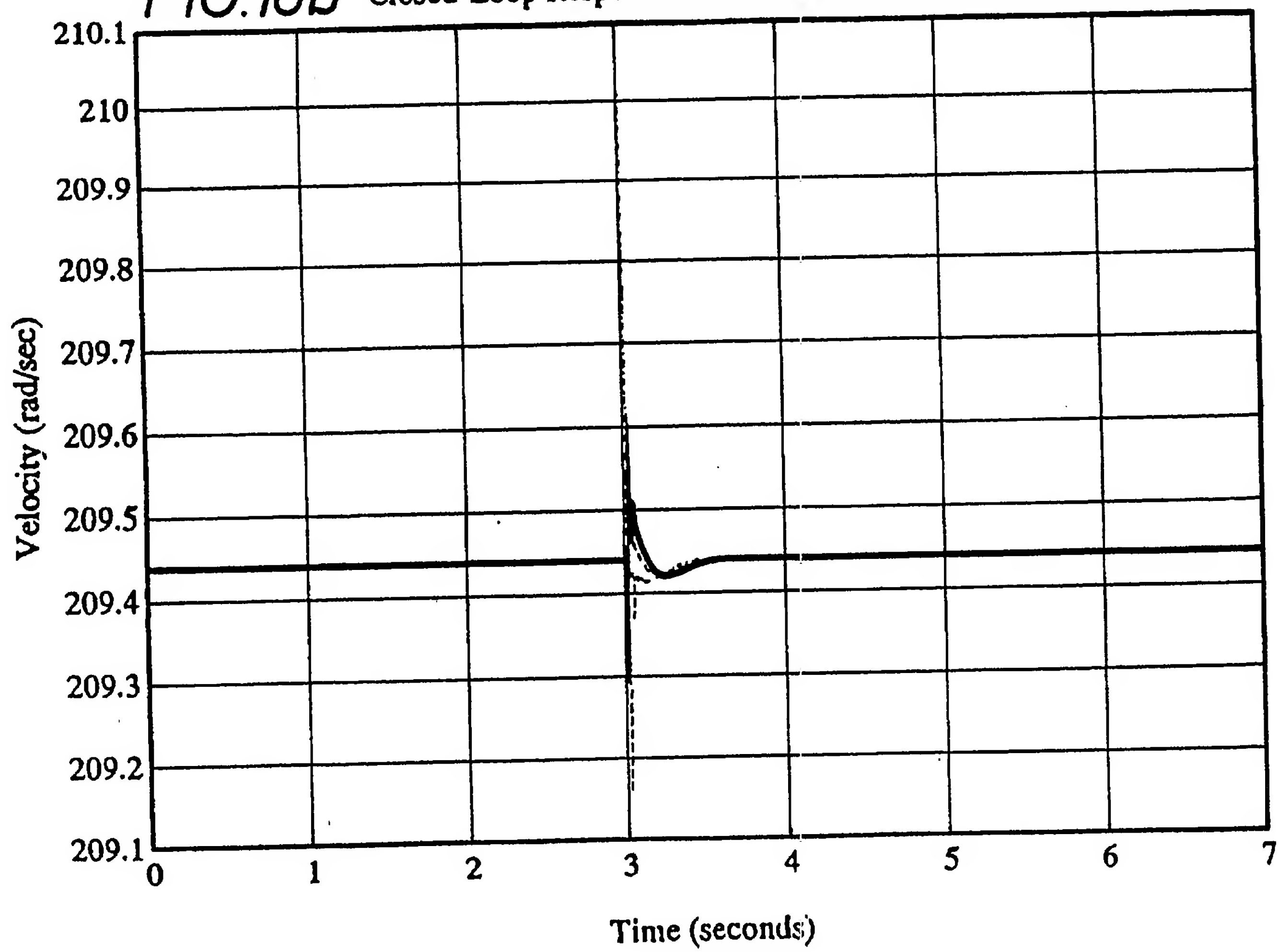

Road Disturbance
Torque (Nm)
200
150
100
50
0
-50
-100
-150
Time (seconds)
0
1
2
3
4
5
6
7

FIG.10b Closed-Loop Response of Driveline Inertias to Road Disturbance



OSCILLATION REDUCTION

This invention relates to an oscillation reduction system. The invention is particularly applicable to an oscillation reduction system for a motor vehicle.

A vehicle driveline consists of components which are arranged to transmit torque from an engine power source to road wheels. Typically, as well as the engine, the driveline will consist of a flywheel, clutch, gearbox, transmission shaft, differential and axle unit connected to the road wheels. All the components will have inertia and compliance. Compliance may be considered as the inverse of stiffness. It relates to the amount of energy an item will absorb when transmitting power.

When the torque transmitted from the engine changes, i.e. when the driveline undergoes a transient, the modes of oscillation of the combination of components can be excited. These oscillations manifest themselves as objectionable vibration experienced by an occupant of the vehicle. They can also generate large transient loadings on driveline components. These loadings may exceed the normal steady state loads imposed on the components by a considerable margin. Added to this, a reversal of the sense of torque applied to a component can exacerbate the problem. Collectively, these problems associated with the driveline may be referred to as 'driveability' problems.

A sudden variation in torque will have a tendency to excite the driveline to oscillate at all the resonance frequencies it might possess. Simply

demanding full power may not be the best answer in all cases as the ensuing amplitude of vibration may be better avoided. However, it is also necessary to avoid compromising the transient performance of the driveline so the driver of the vehicle is largely unaware that any intermediate control, having a detrimental effect on performance, is applied.

Various attempts to address this problem have been proposed. However, these have largely involved tailoring a predefined torque rise profile to the engine concerned, or using a throttle damper, to avoid sudden torque transitions in the driveline.

The problem with these approaches has been that they are all essentially open loop considerations that take no account of the actual instantaneous behaviour of the driveline. Such approaches as predefining the torque rise profile have no ability to take into account driveline oscillations induced by factors external to the driveline itself, e.g. the nature of clutch engagement or irregularities in the road surface.

It is an object of the present invention to provide a oscillation reduction control system for a driveline which is based on a predictive model of the oscillation response of the driveline.

According to the present invention there is provided an oscillation reducing engine speed control system for a power transmission driveline including an engine arranged to transmit power via the driveline to a power output, the system comprising: a control unit programmed with oscillation reducing engine speed control functions based on an oscillation model of the driveline; means responsive to an output of the control

unit for adjusting the power applied to the driveline from the engine, the control means being operable in a control period to produce an engine speed control function, constituting the output from the control means, in dependence on a predictive oscillation response of the driveline at a given engine speed to minimise the oscillation of the driveline.

Preferably, the system includes at least first sensing means operable to provide a component speed and/or relative displacement input to the control unit, the control unit being arranged to monitor the output of the sensing means and to compare an instantaneous engine speed predicted by the control unit according to the oscillation model with an actual speed of the component at that instant. The control unit may then modify its control output on the basis of its comparison in the event that the anticipated effect of the control function is substantially at variance with the output of the sensing means.

The relative instantaneous speed and displacement with respect to a reference speed and displacement can be used to minimise the velocity and displacement differences between different parts of the driveline which arise during driveline oscillation. It should be noted that in a compliant driveline relative displacements between component parts will exist whenever the driveline carries a torque. However, these remain constant for a given torque and are not a constituent of vibration. It is transient changes in relative displacement which can be objectionable.

It is further preferable to provide the control unit with as much information on the relative positions of driveline components. However, practically there is

a limit on the level of monitoring that should be carried out, both in terms of the number of sensors to be read by the control unit and the convenience and expense of mounting a multiplicity of sensing devices in relation to the driveline components.

Preferably, the first speed sensor is arranged to sense the speed and/or position of a flywheel part of the driveline. A second sensor may also be provided to sense the speed and/or position of an output of the driveline, e.g. a road wheel on a vehicle. These are convenient because they are indications commonly taken in motor vehicles having an engine management system and an anti-lock braking system, respectively. Alternatively where the driveline consists of a component such as a transmission shaft, shaft encoders may be usefully employed to sense velocity and position information with respect to one position on the shaft or possibly at spaced positions in order to provide a relative reading of the shaft twist.

A control system according to the invention modulates the engine torque output in response to the velocity and/or displacement of several points on the driveline to achieve minimised driveline component oscillations while maintaining a transient response to speed increase/decrease demand signals.

The present invention can be put into practice in several ways one of which will now be described by way of example with reference to the accompanying drawings, in which:

Figure 1 is a schematic block diagram of a motor vehicle closed loop speed controller according to the invention;

Figures 2 to 6 are graphs of torque input and

driveline velocity output responses of a motor vehicle driveline; and

Figures 7 to 10 are graphs illustrating the response of a closed loop speed controller according to Figure 1.

Referring to Figure 1, a closed loop motor vehicle driveline speed controller comprises an engine management unit (EMU) 10 which has an output arranged to control the speed of a vehicle internal combustion engine 12 in known manner. A driver of the vehicle issues a demand signal which is input to the EMU as a set point signal 14. The demand signal may be related to a desired speed for the vehicle, a desired torque output from the motor, a combination of the two, or some other desired output.

Other inputs to the EMU include the output of a flywheel speed sensor 16 which is a conventional sensor, such as a Hall effect device, arranged to transmit signals indicative of the speed of the engine by picking up the rotation of the flywheel starter motor engagement teeth. This provides a direct reading of the engine speed. A further input to the EMU is from the output of an anti-locking brake system (ABS) speed sensor 18 which is arranged to sense the speed of rotation of the road wheel 20 of the vehicle.

The engine is part of a power train or driveline 22 to the drive wheels. Typically, on a rear wheel drive vehicle, the driveline will include a clutch, gearbox, transmission shaft, differential and axle unit. Of course, on a front or four wheel drive vehicle the driveline will differ from or comprise additional components to this, as will be apparent to the skilled person.

The driveline, including the engine, and the rest of the vehicle will have characteristic inertia and resonance frequencies which can be modelled according to conventional control engineering theory.

Representing the relative positions and velocities of the various driveline components by a vector $\underline{x}$, the response of the driveline to a torque input u can be described by the equation:

$$\underline{x} = A.\underline{x} + B.u$$

The eigenvalues and eigenvectors of the matrix A describe the frequencies and modes of oscillation of the system as a whole. From the point of view of the engine and driveline, the vehicle inertia can be regarded as an effective rotational inertia attached to the outside of the vehicle tyre. The tyre can be reasonably disregarded as being very stiff. Thus, the vehicle inertia can then be added to the wheel rotational inertia. Alternatively, the vibrational model of the system may be expanded to include the tyre as a separate item.

A controller can be designed equivalent to:

$$u = K.(\underline{x}-\underline{r})$$

to modify this response. The vector $\underline{r}$ is the desired value or set point of $\underline{x}$. Different control design approaches will give rise to different choices of the controller matrix K and hence to subtle differences in behaviour of the resulting controlled systems. Different control designs will achieve different control objectives, and thus can give noticeably different responses. For example, a particular control design technique would enable weighting of the errors at different frequencies by different amounts. Whereas, a design technique could place equal emphasis

on an error which is oscillating with a high frequency and an error at a low frequency. The high frequency errors may require extreme control actions compared to the low frequency errors, but it is more likely to want to design a controller which would not attempt to control the high frequency errors because of the level of control actions required even if such high frequency errors are practicably controllable at all. It is possible to design a controller which does not attempt to suppress the low frequency errors. Alternatively, certain frequencies may have particular annoyance value and require particularly strong suppression.

Optimal controller design minimises a defined 'cost function' which is expressed as a formula defining the performance of the controller. Typically, this is expressed in terms of the sum of the squares of the errors between a number of variables in the controlled system and their set point values. By choosing different cost functions, control systems can be designed to achieve a wide variety of performance criteria objectives. In relation to this invention the objective may be considered as minimising the velocity differences between different parts of the driveline which arise during driveline oscillations. Constant relative displacement of a point in the driveline relative to another point in the driveline will be a factor in a compliant system because of the application of, e.g. torque. The objectionable vibration is produced by transient changes in velocity/displacement superimposed on that constant offset induced by the applied torque. It is the transient velocity/position which must be reduced or avoided.

Another way of viewing the objective is that, when

the driveline is accelerated, it should be accelerated as a whole - not the front bit first with the rear end catching up later. If one part of the driveline gets disturbed (accelerated) then we wish to bring it 'back into line' as quickly as possible.

As a practical matter, the cost function should also include a term which is related to the torque required by the controller.

The cost function associated with the states (positions and velocities) of the system may be of the form:

$$J = q_i . x_i^2$$

where q_i is the weighting associated with the state x_i. Minimising this cost function has the effect of minimising the corresponding value of the state. The states with higher weights will be minimised preferentially.

We can add a term to this cost function,

$$J = q_i . x_i^2 + r . u^2$$

Now large values for the control action will also be penalised, and the higher the value of r, the smaller the control actions which will be demanded by the controller designed using this cost function.

In practice, differences between states are weighted, or differences between states and their steady-state values.

The cost function will be designed to prevent the controller from demanding unachievable torques in attempts to obtain a fast, accurate and oscillation-free response of the driveline.

The present invention relies on a oscillation model of the driveline that is developed according to control theory based on the notion that the vibration

characteristics can be interpreted from the displacements of the driveline components relative to one another. The transient shift in relative positions is the source of the vibration. These relative positions can be predicted using control theory for each component according to the torque applied by the motor. If there is a perfect model and perfect knowledge of the disturbances, then it is possible to make perfect estimates of the output. In order to control the system we may need to know the values of the internal states of the system. These can also be estimated from our perfect model. However, in practice our model is not perfect, and the system is subject to disturbances that cannot be measured. This will result in poor estimates of the control output, and more importantly poor estimates of the internal state which would, in turn, lead to inappropriate control actions being made.

This poor estimate, caused by the modelling errors and unknown disturbances, can be improved by comparing our estimated output with the measured output. Any discrepancy we can use to modify the model predictions to bring them back on course.

Thus, the open-loop system would work well with a perfect model and complete knowledge of the disturbances (the simplest case being that there are no disturbances). Usually, however, the model is not completely accurate, and there are many unmeasurable disturbances to the system. Thus measurements of the system's response are very useful for improving knowledge of the internal states of the system. Furthermore, the more measurements made the better the estimate of the internal states can be (especially

given the fact that any one measurement can be subject to measurement errors).

In a closed loop form of the invention the actual displacement of a component due to oscillation can be monitored directly and used as a confirmation of the vibrational model which is used to predict the position as a result of the frequency of oscillation.

A vibrating system, such as the driveline, contains stiffnesses and dampings. With these elements the 'parameters' or variables it is necessary to know, in order to predict the behaviour of the system, are the position and velocities of the various parts of the system. This is because it is these positions and velocities which appear in the equations which govern the behaviour of the system. These positions and velocities are called the <u>states</u> of the system previously mentioned, and the model is based on these.

With such a controller in operation the accelerator pedal position ceases to have a direct correspondence to the throttle position. What significance is then attached to accelerator position is an arbitrary decision and could be programmed in response to driver preference, e.g. accelerator position relates to desired speed or desired acceleration. A close similarity to the conventional indication by pedal position is possible if it is arranged that it represents the desired torque at the road wheels. Because of the ability of the engine and transmission to vary the response to develop a desired torque, the pedal output can be scaled according to gear ratio.

In a gasoline engine, the accelerator pedal is linked to the throttle, which is the butterfly valve

(or similar) which restricts the air flow into the engine. In a diesel engine there is no throttle, and the accelerator pedal is then linked to a 'demand level' on a governor which changes the fuelling to the engine according to demand lever position and engine speed.

Thus, the accelerator pedal position might be related to road wheel torque. In a conventional gasoline engine, there is a good correspondence between pedal position and engine torque - engine speed does affect this relationship, but at any speed greater depression of the pedal equals more torque. The actual torque at the road wheels may oscillate relative to the engine torque due to the vibration effects.

To control road wheel torque to reflect pedal position, in a high gear maximum engine torque will produce less torque at the road wheels than in a low gear. Thus, the controller should take account of this - the driver used to a conventional vehicle would not expect as much road wheel torque in a high gear, as a low gear, for a given pedal position.

The basic open loop embodiment of the invention comprises a full vibrational model of the driveline in the EMU which is arranged to receive a speed or other demand signal from the driver and to initiate a torque control output that drives the engine through the change in speed with minimal oscillation.

In a practical situation it will be the model inaccuracies and unmeasured disturbances which would affect the system, particularly an open-loop system. A further consideration will be that, in each gear, the apparent inertia of the back-end of the driveline, as seen by the engine, will be different. Thus, a

different model is required for each gear ratio.

In a closed loop embodiment the same control regime applies. However, the speed sensors are used to provide readings of the instantaneous speeds of driveline components as an indication of oscillation. These readings are used to confirm the predicted control model theory. In the event of a disparity the control model oscillation predicted is modified accordingly.

In practical systems, it is not possible to measure the velocity and displacement of all the driveline states specified in the state vector $\underline{x}$ of the vibrational model. However, by using a state estimator and by knowing the torque which is being demanded by the controller, this limitation can be overcome. A state estimator is a known control technique whereby values of all the driveline component relative positions and velocities can be estimated from those measurements which can be made.

The control loop of Figure 1 illustrates the use of the commonly existing flywheel and wheel speed sensors. Thus, using a state estimator a controller having all the benefits of state feedback can be realised with measurement of only these two variables.

Engine torque can be modulated in various ways. In a gasoline engine large torque variations are possible by putting the throttle under the direct control of the controller. The limitation of this approach is that engine torque cannot be changed rapidly enough due to the relatively high inertia of the air and other gases flowing through the engine. Thus, throttle control tends not to be sufficiently responsive initially. More rapid changes in engine

torque may be achieved by altering the spark ignition timing, although the amount of torque variation achievable by this is usually limited. A combination of both techniques offers the necessary wide torque range and rapid initial response.

In a diesel engine, torque output can be rapidly modulated by adjusting the quantity of fuel injected into each cylinder. This causes a negligible change in air/gas flow through the engine.

Figure 2 shows the uncontrolled response of the driveline velocities to a step change in engine torque. Figures 3 and 4 show the effect of reducing the rate at which engine torque increases, such as would be obtained by adding a throttle damper. Driveline oscillations are reduced but the system responds more slowly and is still prone to oscillations.

Figure 5 shows the response of the uncontrolled driveline to a poor or snap clutch engagement. No change in torque is demanded at the instant of engagement, and so the system has no ability to affect this transient. Figure 6 shows the uncontrolled response of the driveline to a disturbance at the road wheels such as might be caused by an irregularity in the road surface. In both cases the driveline is subjected to considerable ringing as a result of the externally applied purterbation in torque.

Figures 7 and 8 show the response of the closed loop optimal controller according to the invention along with the velocity set point profile. In Figure 7 it can be seen that significant reductions in driveline oscillation are achieved with minimal impact on the speed of response of the whole driveline. The tradeoff between the speed of response of the driveline and the

severity of driveline oscillations can be systematically altered by the choice of the state and input 'weightings' used during the controller design. The result of using a controller design to achieve a faster speed of response is shown in Figure 8. Having defined a set of weightings for the states, and for the inputs, a number of matrices and coefficients are derivable, (often referred to as A, B, C, D, Q and R) which represent these weightings and the system model. Algorithms are then used which take these quantities as input, and yield the coefficients of the chosen controller design as an output.

Figure 9 shows the response of the closed loop optimally controlled system to the snap clutch engagement in Figure 5. Figure 10 shows the responses to the road disturbance. It can be seen that the controller will act to minimise driveline oscillations whatever the cause whereas the unmodulated approach has no effect if the response time of the controller is to be substantially maintained.

The use of alternative control design techniques will enable different control system behaviour which will still achieve the objectives described above. In particular, the controller may be designed so as to demand only torques which can be practically achieved with a given engine. Alternatively, oscillations within a frequency range which corresponds with vehicle body resonance or high subjective annoyance could be minimised preferentially.

While the present invention has been described in relation to motor vehicles and internal combustion engines, the invention is also equally applicable to other situations in which engines or motors as sources

of power such as, for example, electric motors are used in a driveline to provide a power output.

CLAIMS:

1. An oscillation reducing engine speed control system for a power transmission driveline including an engine arranged to transmit power via the driveline to a power output, the system comprising: a control unit programmed with oscillation reducing engine speed control functions based on an oscillation model of the driveline; means responsive to an output of the control unit for adjusting the power applied to the driveline from the engine, the control means being operable in a control period to produce an engine speed control function, constituting the output from the control means, in dependence on a predictive oscillation response of the driveline at a given engine speed to minimise the oscillation of the driveline.

2. A system as claimed in claim 1 including at least first sensing means operable to provide a component speed and/or relative displacement input to the control unit, the control unit being arranged to monitor the output of the first sensing means and to compare an instantaneous engine speed or position predicted by the control unit according to the oscillation model with an actual speed of the component at that instant.

3. A system as claimed in claim 2 in which the control unit is operable to modify its control output on the basis of said comparison of the instantaneous speed predicted by the control unit according to the oscillation model with the actual speed of the component at that instant in the event that the anticipated effect of the control function is

substantially at variance with the output of the speed sensing means.

4. A system as claimed in claim 2 or 3 in which the first speed sensing means is arranged to senser speed and/or position of a flywheel part of a driveline.

5. A system as claimed in any of claims 2 to 4 including a second speed sensing means arranged to senser speed and/or position or an output of the driveline.

6. A system as claimed in any of claims 2 to 5 in which the driveline comprises a transmission shaft, the speed sensing means comprising shaft encoders arranged to sense velocity and position of the transmission shaft with respect to a predetermined position on the shaft or as a differential between two spaced sensing means on the shaft.

7. A motor vehicle having a driveline for transmitting motive power from an engine to road wheels constituting the power output, the vehicle including a system as claimed in any preceding claim.

Patents Act 1977
[ ]aminer's report to the Comptroller under Section 17 (The Search Report)

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Relevant Technical fields

(i) UK Cl (Edition L) G3N (NGE1B,NGE1A,NGE1,NGBE1, NGBX,NGD,NG1B)

(ii) Int Cl (Edition 5) G05B, G05D, H02P, F02D

Databases (see over)

(i) UK Patent Office

(ii) ONLINE DATABASES: WPI

Searcn Examiner

MR A BARTLETT

Date of Search

15 FEBRUARY 1993

Documents considered relevant following a search in respect of claims 1-7

Category (see over)	Identity of document and relevant passages	Relevant to claim(s)
X	EP 0226965 A2 (EATON CORP) see page 2 lines 26-45 and page 4 lines 14-50 in particular	1-4 & 7 at least
A,E	WO 92/14296 (YASKAWA) (5 February 1991) see Abstract	1 at least

Category	Identity of document and relevant passages	Relevant to claim(s)

Categories of documents

X: Document indicating lack of novelty or of inventive step.

Y: Document indicating lack of inventive step if combined with one or more other documents of the same category.

A: Document indicating technological background and/or state of the art.

P: Document published on or after the declared priority date but before the filing date of the present application.

E: Patent document published on or after, but with priority date earlier than, the filing date of the present application.

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Databases: The UK Patent Office database comprises classified collections of GB, EP, WO and US patent specifications as outlined periodically in the Official Journal (Patents). The on-line databases considered for search are also listed periodically in the Official Journal (Patents).

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